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Reducing water usage with rotary regenerative gas/gas heat exchangers in natural gas-fired power plants with post-combustion carbon capture

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ABSTRACT

It is possible to greatly mitigate the increase of water usage associated with the addition of carbon capture to fossil fuel power generation. This article presents a first-of-a-kind feasibility study of a series of technology options with rotary regenerative gas/gas heat exchangers for the management of the water balance around post-combustion carbon capture process integrated with CCGT (Combined Cycle Gas Turbine) plants with and without EGR (exhaust gas recirculation). Hybrid cooling configurations with a gas/gas heat exchanger upstream of the direct contact cooler reduce cooling and process water demand by 67% and 35% respectively compared to a wet system where the flue gas is primarily cooled prior to the absorber in larger direct contact coolers. The CO_2-depleted gas stream is then reheated above 70 °C with enough buoyancy to rise through the stack. Dry air-cooled configurations, relying on ambient air as the cooling medium, eliminate the use of process and cooling water prior to the absorber and the temperature of the flue gas entering the absorber is unchanged. Rotary regenerative heat exchangers do not introduce significant additional pressure drop and gas leakage from a high CO_2 concentration stream to a stream with lower concentration can be minimized to acceptable levels with available strategies using a purge and a scavenging slipstream from the higher pressure flow.

1. Introduction

Electricity generation from thermal power plants requires the use of significant quantities of water, primarily for the purpose of condensation of the steam in the condenser of the steam turbines. Most of the cooling capacity is provided by water diverted from surface or ground water sources such as, rivers, tidal estuaries and coasts. This is generally referred to as water abstraction. In industrialised countries, water abstractions for electricity production can constitute up to around 40% of the total abstraction from fresh water source. This is notably the case in Europe [1,2] and the United States [3,4]. The fraction of the water abstracted that is permanently withdrawn from its source and is no longer available is referred to as water consumption. It has evaporated and is removed from the immediate water environment.

Water is becoming a scarce natural resource and freshwater and marine environments are under increasing pressure, mainly because of growing populations, changing socioeconomic conditions and climate change [5]. Moreover, water demand is expected to rise further in places due to electrification with new-build thermal power plants and, even more, if decarbonisation of electricity generation takes place with high contribution of carbon capture and storage technologies in fossil fuel-fired plants. Novel engineering solutions are necessary to limit the additional water abstraction and water consumption of power plants with CCS [6].

The availability of cooling water might constitute a limitation for the full scale deployment of carbon capture and storage, particularly in regions with increasingly restricted access to cooling water and limited availability of fresh or sea water abstraction licences, as indicated by Ref. [5]. Restricted access to water is notably likely to be the result of growing concerns to protect inland water resources and minimise marine ecological impact, or simply seasonality change in cooling water availability and fresh water shortages. In periods of low flow, thermal plants may be required to shut down.

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This article examines novel technology options with rotary regenerative gas/gas heat exchangers for water reduction in CCGT (Combined Cycle Gas Turbine) plants with PCC (post-combustion capture), and then expands the analysis to CCGT plants with EGR (Exhaust Gas Recycling), a possible way of increasing CO_2 concentration in exhaust flue gas for capital cost reduction. A range of configurations are examined for water reduction with hybrid and dry air cooling of the flue gas streams entering the absorber in the post-combustion capture plant and provide the necessary heating of the CO_2-depleted flue gas leaving the absorber, prior to being released in the power plant stack.

2. Heat and water management of flue gas streams in post-combustion carbon capture scrubbing technology

Post-combustion carbon dioxide capture using amine-based solvent technology is a possible option for commercial scale deployment of CCS, e.g. the Boundary Dam CCS Project in Canada in operation since September 2014, or the Peterhead CCS project in the UK in a FEED (front end engineering design) phase at the time of writing. Currently available technologies require cooling capacity for the exhaust flue gas entering the absorber column, water wash system, absorber intercooler, lean solvent cooler, reflux reboiler duty and CO_2 compressor interstage coolers. Unless dry cooling systems are used, this drastically increases the cooling water demand if carbon capture were to be added to a thermal power plant with either a once-through cooling system or a recirculating cooling system with evaporative cooling tower [2–5,7].

In CCGT plants without CO_2 capture, flue gas typically leaves the HRSG (heat recovery steam generator) at a temperature ranging from 80 to 150 °C. The exhaust flue gas temperature is sufficiently high to rise through the stack by buoyancy effects and meet the regulatory requirements of stack plume visibility that exist in some jurisdictions. When post-combustion carbon capture using amine-based solvents is implemented, the temperature of the flue gas entering the absorber has to be adapted to the specific solvent. The selection is generally a trade-off between kinetics and capacity to enhance the absorption process. The exhaust flue gas is typically cooled down by means of a DCC (direct contact cooler) until it reaches an optimal temperature, which has been reported to be between 35 °C [8,9] and 40 °C [10–13]. Temperatures as high as 50 °C have also been presented in pilot plant data [14]. A large volume of process and cooling water is therefore required in the direct contact cooler.

At the outlet of the absorber, the CO_2-depleted gas typically leaves at a temperature similar to the inlet flue gas. A water wash section is located at the top of the absorber to avoid solvent vaporization and solvent losses into the atmosphere and to maintain the water balance of the system. The CO_2-depleted gas is then either directly released into the atmosphere through a wet stack, e.g. in Refs. [15], or must be reheated in order to increase buoyancy forces and avoid plume visibility in a dry stack design, e.g. in Ref. [9]. The temperature of the gas stream entering the stack depends on the requirements for plume rise and dispersion, driven by the national/regional legislation and geographical location of the plant.

A heat integration option transferring heat from the gas stream entering the absorber to the gas stream leaving the absorber has the advantage of reducing or eliminating process and cooling water requirements of the direct contact cooler, and ensuring that the flue gas reaches an appropriate temperature at the stack. This can be achieved with the use of a rotary regenerative gas/gas heat exchanger; a technology widely used in coal fired power plants to increase boiler efficiency.

An analysis of thermal performance is conducted for two configurations of gas-fired power plants:

- A conventional (air-based combustion) CCGT power plant with post-combustion capture
- A CCGT power plant with post-combustion capture using Exhaust Gas Recirculation.

Hybrid and dry cooling systems are proposed to reduce the size or eliminate the direct contact cooler upstream of the CO_2 absorber. Different configurations of rotary regenerative gas/gas heat exchangers are examined with respect to water usage, stack temperature and their respective power requirements and design parameters related to their geometry and operational conditions.

EGR (Exhaust Gas Recirculation) is the process of diverting a fraction of the combustion gas at the outlet of the HRSG to the inlet of the compressor of the gas turbine. It has been widely studied in literature as a strategy to increase the CO_2 concentration and reduce the volumetric flow rate of the flue gas to feed to the absorber of CCGT plants with post-combustion carbon capture [16]. Greater CO_2 partial pressure enhances the absorption process and the lower volume of the gas to be treated reduces the size of the absorber tower of the carbon capture plant [17]. The recirculated gas stream is typically cooled down before it is mixed with ambient air, as lower inlet temperature favours compression and increases gas turbine power output. In previous work, little consideration has been given to optimising heat and water management of the flue gas loop. The same cooling equipment is typically used for both streams [18,19]. Recent studies have, however, proposed the recycling of exhaust gas at lower temperatures, with two direct contact coolers of smaller size introduced to the process [9].

3. Rotary regenerative gas/gas heat exchanger technology

Rotary gas/gas heat exchangers rely on a technology widely used in coal fired power plants to pre-heat the primary and secondary combustion air streams using the flue gas stream exiting the boiler. The same technology is also used to recover heat from the inlet stream of FGD (flue gas desulphurization) units to increase the temperature of the outlet stream and its buoyancy at the inlet of the stack [20]. The concept of regenerative heat transfer can be easily adapted to CCGT with CO_2 capture applications.

In rotary regenerative heaters, heat is indirectly transferred by convection as a heat storage medium is periodically exposed to hot and cold gas streams, flowing in a counter current arrangement. Continuous cycling exposure is accomplished by a rotary mechanism and steel elements are used as the heat storage medium. Regenerative heating surface elements are a compact arrangement of pairs of specially formed metal plates, providing very high surface area per unit volume. Each element pair consists of a combination of flat, notched, corrugated, herringbone or undulated plate profiles which are packed into self-contained baskets installed into the rotor in two or more layers [20]. The main components in the rotary heat exchanger design are illustrated in Fig. 1. Based on the number of streams involved in the heat transfer process, a bisector, trisector or quadsector configuration can be selected and the open cross section is divided among the gas streams, proportionally to their mass flow rate, so that the gas velocity at either side of the sector plate is similar and does not result in high pressure drops. Due to the rotation, leakage occurs between the gas streams. Direct leakage (also called gap leakage) occurs through the radial, axial and other seals between rotating and stationary parts, due to pressure differential between the gas streams on either side of the sector plate. Multiple axial and radial seals under the sector plate are used to lower direct leakage flow. Entrained leakage (also called carryover leakage) is the gas contained in a rotor sector as it is carried into the other stream by rotor rotation. This leakage is directly proportional to the void volume of the rotor and the rotor

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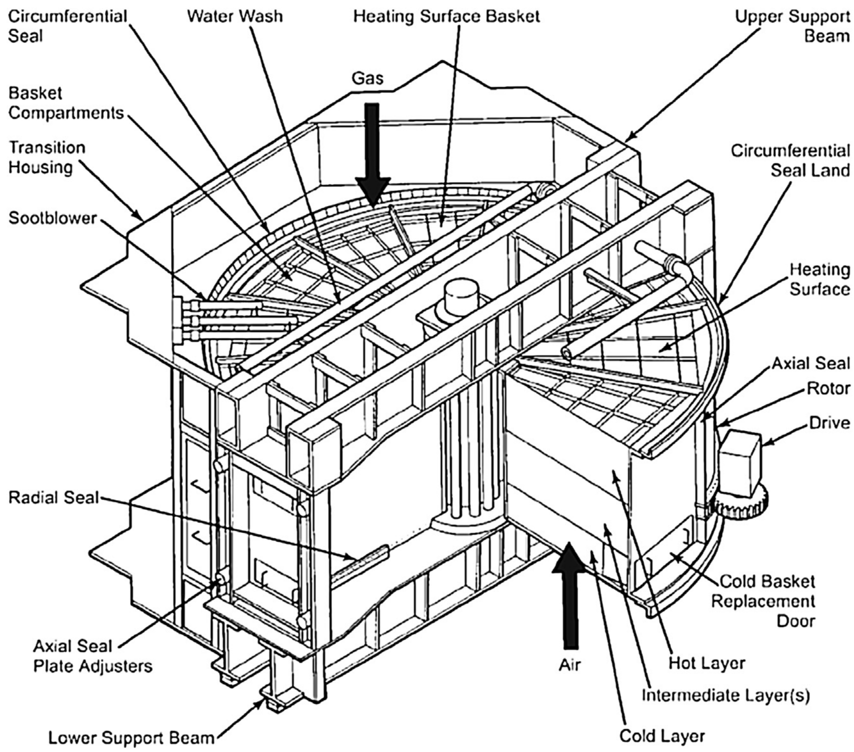

Fig. 1. Example of a typical rotary regenerative heat exchanger used in coal-fired power plants [20].

speed. Regenerative rotary heat exchangers are therefore designed to meet thermal performance requirements, while maintaining the leakage levels and pressure drop as low as possible.

4. Methodology for the selection of regenerative rotary gas/gas heat exchangers

The sizing of the rotary heat exchangers and the thermal performance analysis uses a methodology specifically developed for this novel application and based on proprietary software developed by Howden Group, a manufacturer of rotary heat exchangers. The configuration dimensions and operational parameters are designed to achieve satisfactory outlet temperature of the gas stream at the cold and hot ends of the device.

The heating surface area is defined by selecting the diameter of the rotor and the length of the metal elements. Energy balance and heat transfer equations are solved to obtain the gas and metal temperature profiles, where the overall heat transfer coefficient is evaluated from correlations that include parameters based on Howden's experience.

The diameter and the length of the metal elements are varied in the analysis, with material specifications, metal thickness and rotor speed identical for all configurations and selected according to practical experience. The diameter is sized to achieve a typical non-erosive gas velocity, around 15 m/s, and the element depth is then sized. Longer elements results in a larger pressure drop while larger diameters result in a lower gas velocity and a lower pressure drop, although this needs to be balanced against the higher direct and entrained leakage levels. Multiple axial and radial seals are used to minimise direct leakage flow rate. An appropriate design is therefore the result of a trade-off to achieve the required outlet temperature with minimum pressure drop, acceptable leakage levels and heat exchanger size.

Due to the smaller temperature differential at the cold end in this particular application, compared to air preheaters in coal-fired power plants, there is a law of diminishing return for each incremental increase in heat exchanger size. Once a certain size is reached, a large increase in the heating surface available then becomes necessary to obtain a marginal decrease/increase of the outlet temperature of the hot/cold streams.

The pressure drop through the heater is a combination of duct losses at inlet of the rotor, expansions losses at outlet of the rotor and rotor losses and it is proportional to the square of the fluid velocity and fluid density, according to the equations included in Appendix C.

A general equation for direct leakage flow rate is also provided in Appendix C to show the important dependent parameters.

Direct and entrained leakage flow rates are estimated based on pressure difference at both sides of the sector plate, void fraction, the dimensions and geometry of the heat exchanger, and perimeter of sealing and gaps. The methodology used in this work to evaluate leakage relies on previous experience in the application of regenerative rotary heat exchanger in pulverised coal plants, operating at temperature ranging from 100 to 300 °C, i.e. considerably higher than the temperature values of 40–100 °C encountered in this application. Thermal expansion and contraction of the perimeter of sealing and gaps in the range of operating temperatures 40–100 °C can be expected to have a small effect on leakages. Leakage levels around 0.5%–1% can be reasonably expected, although it would be possible to engineer purge and scavenge systems to minimise leakage flow rates even further. The sensitivity analysis in Section 7 shows that these conservative estimates of leakage rates could be tolerated and are likely to be deemed acceptable since they would only have a limited impact on the operation of the post-combustion capture plant.

5. Novel integration options for heat and water management in combined cycle gas turbine plant with post combustion capture.

Three alternative configurations using rotary heat exchangers are compared to the standard configuration of using exclusively a direct contact cooler.

- A hybrid cooling system which includes a rotary regenerative gas/gas heat exchanger upstream of a direct contact cooler. Heat is transferred from the exhaust flue gas leaving the HRSG into the CO_2-depleted gas stream leaving the absorber to enter the stack.
- A dry air-cooled system which consists of two rotary heat exchangers in series. A rotary air/gas heat exchanger replaces the direct contact cooler in the configuration, downstream of a rotary regenerative gas/gas heat exchanger.
- A dry air-cooled system with a more compact arrangement to the previous one, which combines the two rotary heat exchangers into a single regenerative rotary gas/gas/air heat exchanger with a trisector configuration.

Block flow diagrams for each configuration in a convectional CCGT plant and a CCGT plant with EGR are shown in Figs. 2 and 3 respectively.

5.1. Wet cooling system with a direct contact cooler (base case)

The gas is brought into direct contact with process water in a counter current configuration in a packed bed column. Water collected in the cooler sump is passed through a dedicated water-cooled heat exchanger before being returned to the top of the column. Cooling water in the heat exchanger is provided by the primary plant cooling system.

In the rest of this article, process water refers to the amount of water used in a recirculating loop to cool down the flue gas by direct contact. There is either consumption or production of process water that needs to be supplied or extracted. The amount depends on the difference between the absolute humidity of the gas at the inlet and at the outlet of the direct contact cooler. With appropriate consideration a neutral balance of the circulating loop can be achieved [10]. Cooling water for the DCC is taken from the primary cooling system of the power plant and used to cool down the process water. The cooling water is either water abstraction in a once-through cooling system or recirculated water in a recirculating cooling system, contributing then to water consumption in the evaporative cooling towers.

5.2. Hybrid system with a gas/gas heat exchanger and a direct contact cooler in series

In this configuration, a gas/gas heater operates upstream of the direct contact cooler in a bisector arrangement. The metal elements contained in the baskets of the gas/gas heat exchanger are heated in contact with the flue gas leaving the HRSG and are subsequently cooled in contact with the treated CO_2-depleted gas, transferring sensible heat from one gas stream to the other later. The typical temperature of 35–45 °C at the absorber inlet cannot be achieved with the gas/gas heat exchanger alone and further cooling occurs in a direct contact cooler tower with lower process and cooling water requirements.

5.3. Dry system with a gas/gas and an air/gas heat exchangers in series

Two consecutive rotary heat exchangers replace the direct contact cooler of the wet cooling system. The first gas/gas heat exchanger transfers sensible heat from the exhaust flue gas stream to the CO_2-depleted gas stream to reach the temperature necessary for adequate

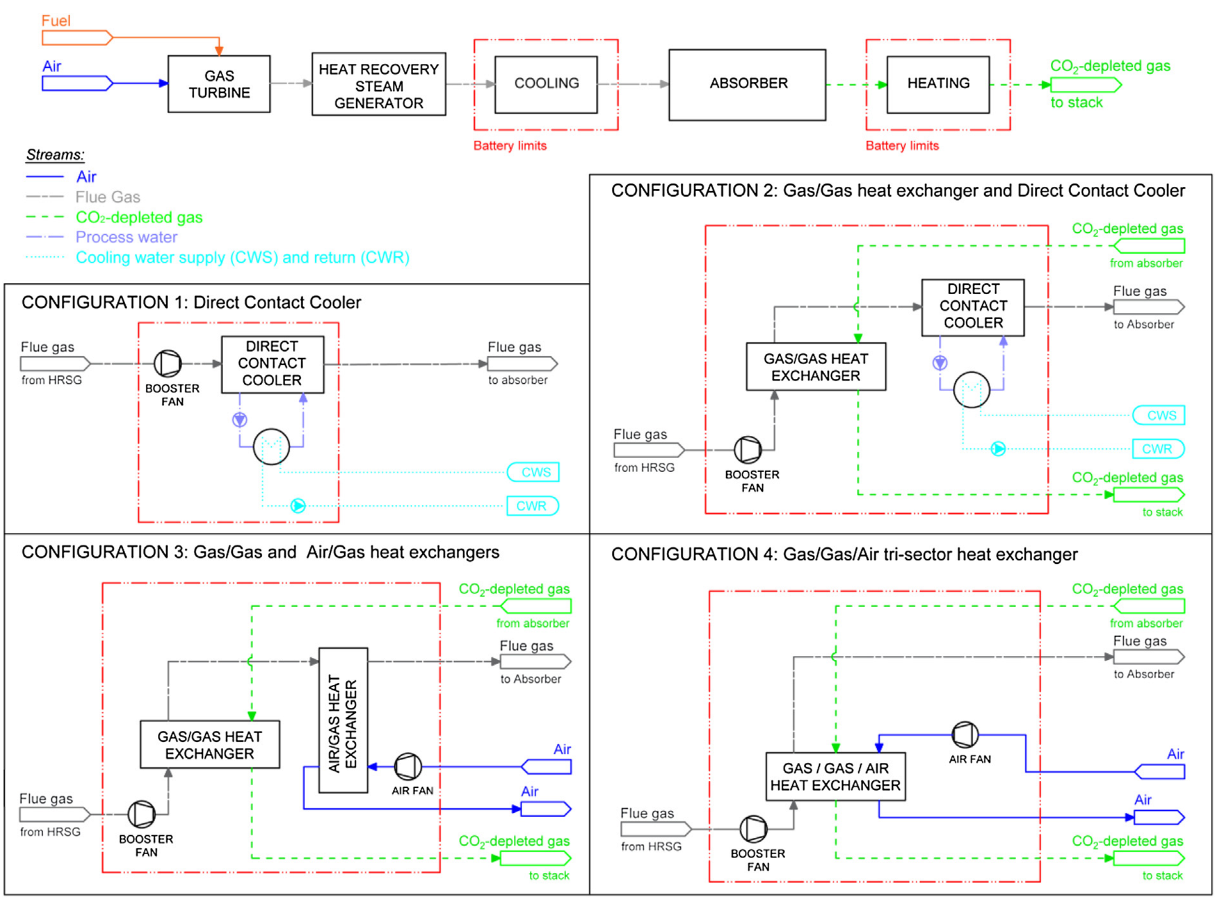

Fig. 2. Block flow diagrams of rotary heat exchangers configurations for a conventional (air-based combustion) gas turbine combined cycle plant with post-combustion carbon capture technology.

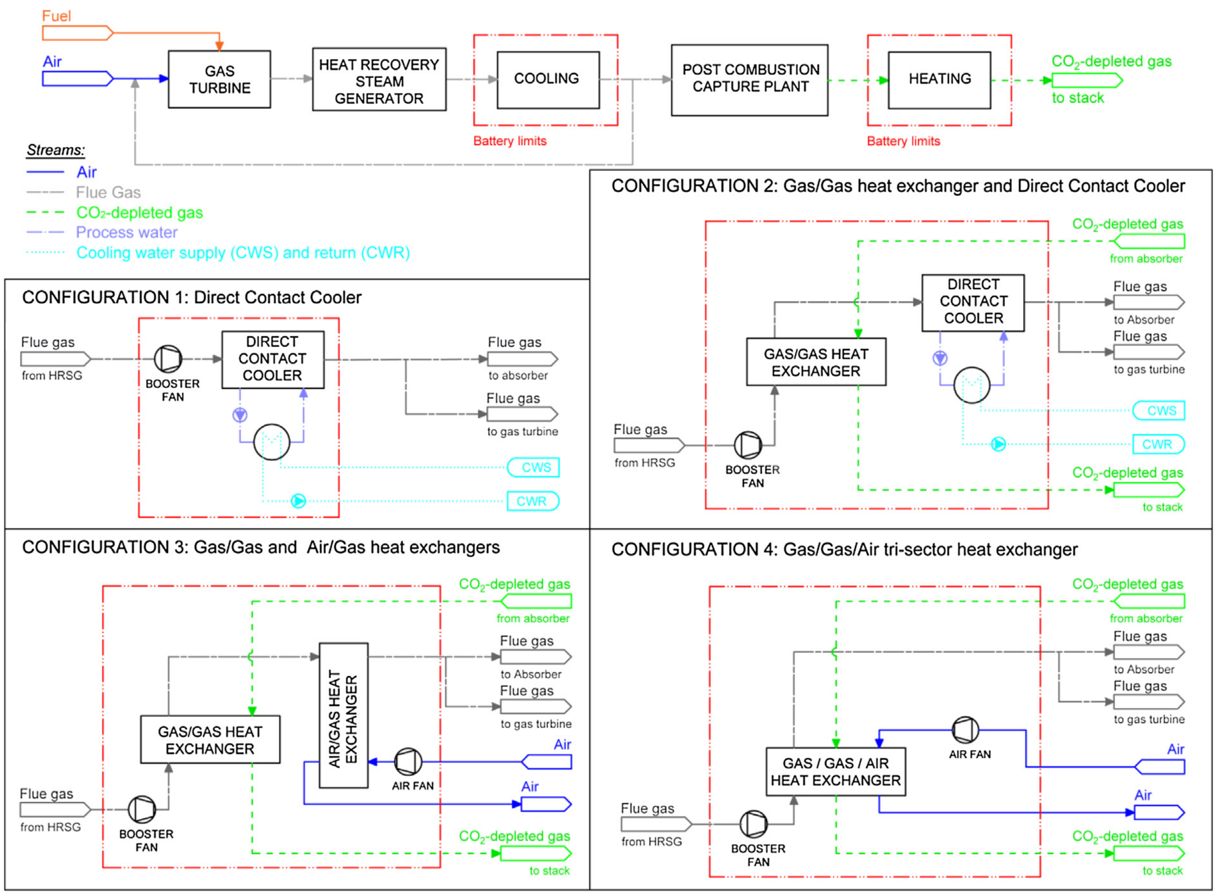

Fig. 3. Block flow diagrams of rotary heat exchangers configurations for a gas turbine combined cycle plant with post-combustion carbon capture technology and exhaust gas recirculation.

buoyancy at the inlet of the stack. The gas/gas heat exchanger design is the same as in the previous configuration. In order to reduce further the exhaust flue gas temperature, an additional rotary air/gas heat exchanger using ambient air as the cooling fluid replaces the smaller direct contact cooler of the hybrid system. A forced draft fan overcomes the pressure drop of the air section of the air/gas heat exchanger.

5.4. Dry system with a trisector gas/gas/air heat exchanger

The two rotary heat exchangers in series of the previous configuration are replaced by a heat exchanger with a tri-sector arrangement with the potential for a large reduction in the overall size, footprint and capital cost. Exhaust flue gas coming from the HRSG flows in a counter current manner to the CO_2-depleted gas stream and to the air stream. The direction of rotation of the heat exchanger is an important consideration so that the baskets turn from the exhaust flue gas section into the absorber outlet gas section and then the air section, as illustrated in Fig. 4. This allows better heat transfer and a smooth temperature gradient of the elements, as well as an adequate direction of leakage with the appropriate pressure gradient, as discussed in the next section.

6. Thermal performance analysis for combined cycle gas turbine plant with air-based combustion and exhaust gas recirculation

The thermal performance analysis of each rotary heat exchanger configuration is evaluated for a flue gas train consisting of one gas turbine, one heat recovery steam generator and one absorber column of a reference CCGT plant with a 2-in-1 configuration (two gas turbines, two HRSGs and one steam cycle). Key values and assumptions are summarised in Appendix A and Appendix B and key findings are discussed below.

For all configurations the gas stream enters the absorber saturated with water at 45 °C and the reheated flue gas enters the stack at a temperature at least higher than 70 °C for good buoyancy

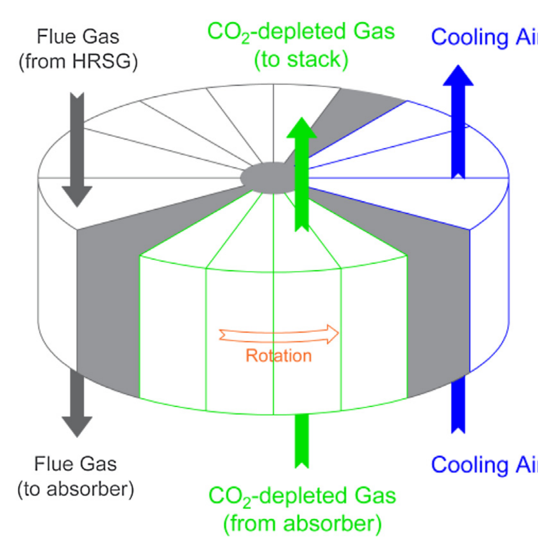

Fig. 4. Direction of the flows in a trisector rotary gas/gas/air heat exchanger.

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effects and plume abatement. The respective dimensions and operational parameters, such as gas velocity, pressure drop, etc. are based on the heating surface area required to reach the outlet temperatures.

6.1. Reference plant

For the reference CCGT plant described in Appendix A, flue gas leaves the HRSG at 86 °C and 101.3 kPa. The booster fan, located upstream of the direct contact cooler, increases the flue gas pressure by around 10 kPa [9], resulting in a temperature rise of around 11 °C. Sensible heat is removed by the direct contact cooler and the flue gas enters the absorber saturated in moisture at the dew point around 45 °C. The CO_2-depleted gas then leaves the water wash section at the top of the absorber saturated in moisture at a temperature of 45 °C, in order to maintain a close to neutral water balance in the absorption tower.

6.2. Thermal performance analysis of regenerative heaters and water management of flue gas streams

Fig. 5 summarises the requirements for process water, cooling water and cooling air for each configuration of rotary heat exchangers for air-based combustion and exhaust gas recirculation CCGT plant at 40% recirculation ratio. Table 1 includes the dimensions of the heating metal elements assembly in the rotary heat exchangers, the temperatures of gas streams and operational parameters.

6.2.1. Combined cycle gas turbine plants with air-based combustion

6.2.1.1. Hybrid cooling systems. In hybrid cooling systems, with a rotary gas/gas heat exchanger and a direct contact cooler in series, heat is transferred in the first device from the gas stream leaving the HRSG to raise the temperature of the gas stream entering the stack above 80 °C, well above the necessary 70 °C for good buoyancy. The cooled gas then leaves the gas/gas heat exchanger at a temperature around 62 °C and is further cooled in a direct contact cooler down to a temperature around 45 °C. The lower amount of heat that needs to be removed from the exhaust flue gas in the direct contact cooler results in a significant reduction in process and cooling water demand of 35% and 67% respectively compared to the traditional wet cooling system with a single direct contact cooler. The dimensions of the baskets containing the heating elements are around 1.22 m long and the selected heater size has a font section area of 152 m^2, which corresponds to a casing diameter of around 14.5 m.

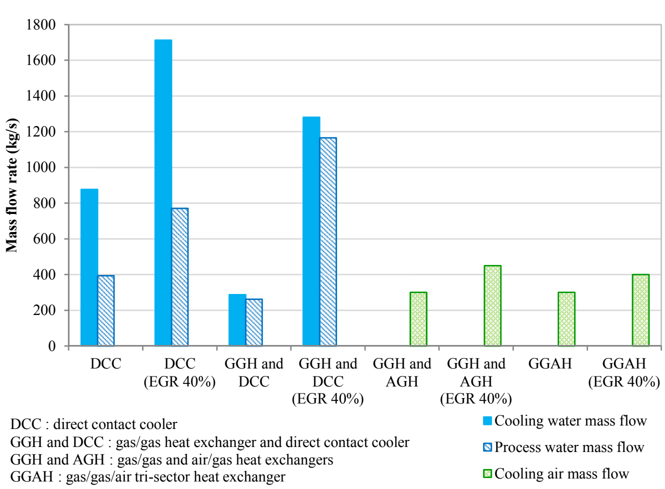

Fig. 5. A comparison of the cooling water, recirculating process water and cooling air mass flow rates per gas turbine train in a convectional (air-based combustion) combined cycle gas turbine plant and a plant with an exhaust gas recirculation ratio of 40%, both with post-combustion capture technology.

6.2.1.2. Dry system with air cooling. In the two dry systems, where air cooling replaces the direct contact cooler, the need for process and cooling water demand for the purpose of cooling the flue gas into the absorber is fully eliminated, contributing to a significant reduction of the overall water usage in the post-combustion capture plant.

- In the configuration with two rotary heat exchangers in series, the air/gas heat exchanger operates with mass flow rate of 300 kg/s of ambient air. The air temperature at the hot end reaches 56 °C. If the hot air and the reheated CO_2-depleted gas streams are mixed, the resulting gas stream enters the stack at around 72 °C. The dimensions of the baskets containing the heating elements are around 1 m long and the selected heater size has a front section area of 125 m^2, which corresponds to a casing diameter of around 13 m.
- The trisector gas/gas/air heat exchanger has the potential for a large reduction in the overall size and capital cost. In addition, the pressure drop on the gas side is reduced from 3.4 kPa to 2.8 kPa. An air mass flow rate of 300 kg/s is heated from ambient temperature, 15 °C, up to around 68 °C. The CO_2-depleted gas stream is reheated up to around 75 °C and enters the stack, after being mixed with the hot air, at around 73 °C. The dimensions of the baskets containing the heating elements are around 1.42 m long and the selected heater size has a front section area of 230 m^2, which corresponds to a casing diameter of around 18 m. When compared to the dimensions for a direct contact cooler treating similar volume of gases [9] of a 10 m × 27 m × 19 m height, this constitute a significant reduction in the overall size.

If a temperature of the gas stream lower than 45 °C at the inlet of the absorber was required, i.e. below the dew point, the temperature difference at the cold end of the rotary regenerative heat exchanger and the driving force for heat transfer would be reduced, leading to an increase in the overall size of the heat transfer surface area. Additional cooling capacity would also be necessary to remove the latent heat of water condensation. This could be achieved either by significantly increasing the heating surface area of the regenerative heater or by operation with larger cooling air flow rates. Water condensation would occur on the heating metal elements in the high temperature gas section and then be transferred to the adjacent section. A large fraction of water droplets could drop by gravity with a carefully selected direction of the gas, and some would evaporate in the air stream. This is illustrated in Fig. 6.

Ambient air conditions could also play an important part if, for instance, the ambient air temperature was lower than the design temperature of 15 °C. The outlet temperature of the gas stream at the cold end would reduce and water condensation would occur. This could however be managed by adjusting the air flow rate. Enamelled heating elements and appropriate material selection would be applied to mitigate any possible long-term corrosion problems, in a similar manner to FGD GGH applications.

6.2.2. CCGT with exhaust gas recirculation: effect of the recirculation ratio

For values of the recirculation ratio comprised from 0 to 40%, water content in the gas stream leaving the HRSG, and the heat transfer duty of the regenerative rotary heat exchangers, increase with the recirculation ratio since the absolute humidity of the

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Table 1
Summary of the results: Rotary heat exchangers basic design and operational parameters (per GT-HSRG-absorber train).

	Gross frontal area	Element depth	Heating surface area	Inlet/Outlet temperature	Hot end velocity	Total pressure drop
	m^2	mm	m^2	°C	m/s	kPa
Conventional air-based combustion Combined Cycle Gas Turbine plants						
Gas/gas heat exchanger (GGH) (with direct contact cooler)	152 (GGH)	1220 (GGH)	66,038	Hot gas 97.5/62.6	13.5	Gas[a] 12.3
				Cold gas 45/82.3	12.9	
Gas/gas heat exchanger (GGH) and air/gas heat exchanger (AGH)	152 (GGH)	1220 (GGH)	66,038	Hot gas 97.5/62.6	13.5	Gas 11.2
				Cold gas 45/82.3	12.9	
	124 (AGH)	1000 (AGH)	45,868	Hot gas 62.6/45	14.6	
				Air 15/56	6.5	Air 0.3
Gas/gas/air tri-sector heat exchanger (GGAH)	230	1420	120,226	Hot gas 97.5/46	11.8	Gas 10.3
				Cold gas 45/75.2	11.8	
				Air 15/68.3	8.4	Air 0.88
Combined Cycle Gas Turbine plants with 40% Exhaust Gas Recirculation						
Gas/gas heat exchanger (GGH) (with direct contact cooler)	152 (GGH)	1220 (GGH)	66,038	Hot gas 97.5/73.7	13	Gas 11.7
				Cold gas 45/91.6	7.1	
Gas/gas heat exchanger (GGH) and air/gas heat exchanger (AGH)	152 (GGH)	1220 (GGH)	66,038	Hot gas 97.5/73.7	13	Gas 10.5
				Cold gas 45/91.6	7.1	
	124 (AGH)	1000 (AGH)	45,868	Cold gas 73.7/49.5	14.5	
				Air 15/57	9.9	Air 0.65
Gas/gas/air tri-sector heat exchanger (GGAH)	230	1420	120,226	Hot gas 97.5/46	11.4	Gas 8.9
				Cold gas 45/80	7.7	
				Air 15/73	11.2	Air 1.2

[a] Pressure drop for the gas side includes the absorber, gas ducts and direct contact cooler and/or rotary heat exchanger. Pressure drop through the absorber, gas ducts and direct contact cooler were assumed to be 5, 2.5 and 2.5 kPa respectively [9]. Pressure drop through the rotary heat exchangers are calculated.

recirculated flue gas stream is greater than that of the ambient air used for combustion.

In addition, the total flow rate entering the regenerative rotary heat exchangers is very similar to air-based combustion, whereas the CO_2-depleted gas flow rate is, on the other side of the heat exchanger, reduced by approximately 40%, at that level of exhaust gas recirculation. The lower cooling capacity is then compensated with larger amount of cooling water or cooling air.

A sensitivity analysis of the recirculation ratio, for the same sizing of the regenerative heat exchangers as for the corresponding configuration of an air-based combustion CCGT plant, shows that the corresponding dew point temperature increases, at constant HRSG outlet temperature for all cases. With 40% EGR, the saturation temperature of the flue gas is around 51 °C, i.e. higher than the 45 °C of a conventional air-based combustion CCGT plant.

One option is to cool the flue gas down to 45 °C to enhance CO_2 removal in the absorber and condense any excess of water. The sensible heat of the gas and latent heat of water have to be removed when the gas is cooled below the dew point. In wet-based and hybrid cooling configurations, process and cooling water flow rates can be increased and excess water purged from the recirculating process water system.

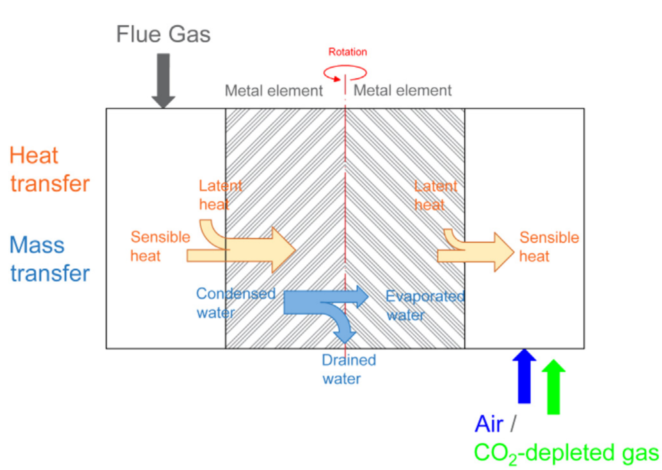

Fig. 6. Heat and mass transfer of water for rotary regenerative heat exchangers operated below the dew point.

For dry systems with air cooling, water condensation will occur on the heating metal elements of the rotary heat exchangers, as shown in Fig. 6. Excessive condensation in the rotary heat exchangers could be avoided with the gas entering the absorber inlet just above the dew point, at expense of the production of water in the water wash section of the absorber tower and the associated complexity of treating the excess water. The condensed water that remains over the metal elements will be partially or totally evaporated increasing the absolute humidity of the air and CO_2-depleted gas at the same time as the streams are heated.

6.2.2.1. Hybrid cooling systems. In hybrid cooling systems, with a rotary gas/gas heat exchanger and a direct contact cooler in series, the temperature of the gas at the cold end of the gas/gas heat exchanger increases, e.g. from around 62 °C for air-based combustion to 74 °C with 40% EGR. The CO_2-depleted gas outlet temperature entering the stack also increases to 90 °C. This is due to the fact that, as explained previously, the full volume of flue gas leaving the HRSG is cooled while only the non-recirculated fraction of the flue gas is reheated on the cold side of the gas/gas heat exchanger.

The larger amount of specific heat and the latent heat of condensation results in significantly higher water requirements in the direct contact cooler compared to the corresponding configuration in an air-based combustion CCGT plant, as indicated in Fig. 5. The absolute increase in process water is higher with EGR than with air-based combustion, since additional latent heat of condensation needs to be removed from the flue gas.

6.2.2.2. Dry cooling systems

- With a gas/gas heat exchanger and an air/gas heat exchanger in series, high levels of EGR lead to higher gas temperature entering the absorber, if the same cooling air flow rate as the air-based combustion case is used. With a cooling air flow rate of 300 kg/s the flue gas is cooled down just above the dew point temperature (around 51 °C) at 40% recirculation ratio. A cooling air flow rate of 450 kg/s is required to further cool the flue gas stream down to 46 °C at the absorber inlet and a temperature of air at the hot end around 57 °C. Fig. 7 illustrates this effect for a range of recirculation ratios, cooling air mass flow and cooling air temperature for the configuration with two rotary heat exchangers in series.

- An increase of the recirculation ratio has a similar effect in the tri-sector gas/gas/air heat exchanger configuration, although to a lower magnitude. The CO_2-depleted gas outlet temperature and the air outlet temperature follow the same trend and increase at larger recirculation ratios, providing a positive effect to enhance stack buoyancy. With an air flow rate of 300 kg/s, at 40% recirculation ratio, the flue gas temperature at the cold end is close to the dew point. A cooling air flow rate of 400 kg/s, as opposed to 450 kg/s, is necessary to cool the flue gas stream down to 46 °C at the absorber inlet. The hot air temperature and the CO_2-depleted gas temperature are respectively 73 °C and 80 °C. Fig. 8 shows the sensitivity of flue gas and air temperature to the recirculation ratios and the mass flow rate of cooling air.

The increase in cooling air mass flow rate with EGR compared to air-based combustion is smaller in magnitude than the rise in cooling water mass flow rate due to the capacity of the air stream for evaporative cooling. The air gains moisture by evaporating condensed water on the metal elements in the untreated flue gas side. The enthalpy of the air increases due to both the increase in temperature and in absolute humidity.

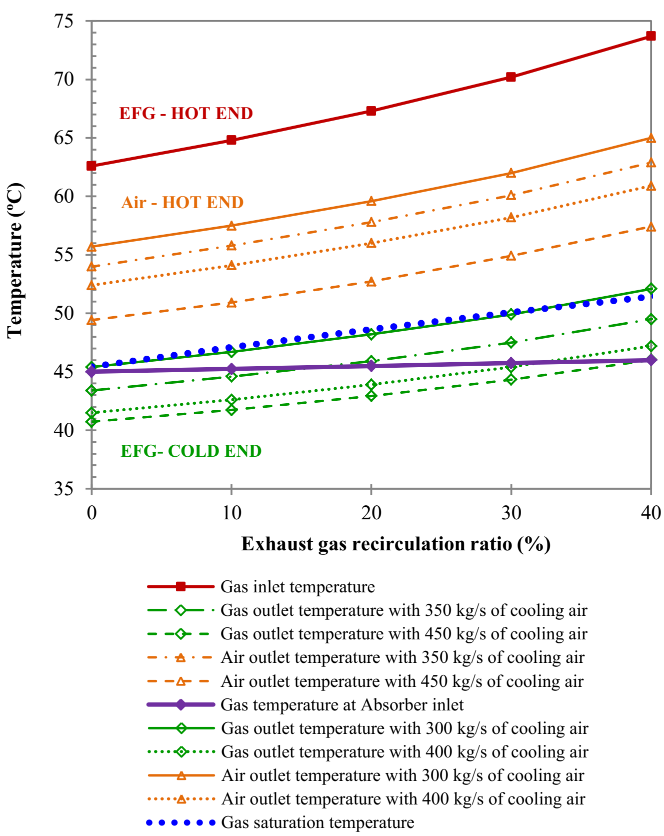

Fig. 7. Dry cooling configuration with two rotary heat exchangers in series: Sensitivity analysis of flue gas and air temperature to the recirculation ratio and the mass flow rate of cooling air.

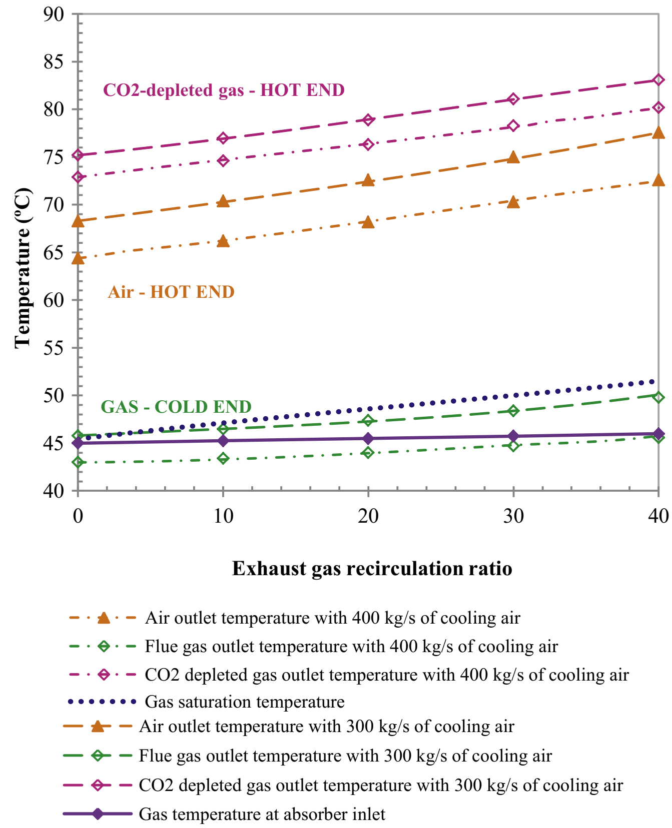

Fig. 8. Dry cooling configuration with a trisector gas/gas/air heat exchangers: Sensitivity analysis of flue gas and air temperature to the recirculation ratio and the mass flow rate of cooling air.

6.3. Effect of the ambient air temperature on the performance dry cooling systems

Dry cooling systems are highly dependent on ambient conditions, namely temperature and relative humidity. A sensitivity analysis is shown in Fig. 9 for a temperature range of ambient air from 10 °C up to 40 °C for dry air-cooled configuration with two rotary heat exchangers in series, initially sized for 15 °C, in a conventional CCGT plant, without exhaust gas recycling.

Fig. 9 illustrates that, for a given heating surface area, an increase in ambient air temperature results in a lower cooling capacity and, therefore, higher outlet temperatures for both the exhaust flue gas and the air streams. At the design air flow rate, the gas outlet temperature is around 50 °C for an ambient air temperature of 30 °C. This can be brought down to 45 °C, the target gas outlet temperature in this work, by increasing the air flow rate from 300 kg/s to 500 kg/s.

By sizing the blower accordingly, larger cooling air flow rates can compensate, to some extent, for the increase in the flue gas outlet temperature. At ambient temperature above 30 °C, an asymptotic behaviour of the air flow rate to maintain gas outlet temperature at 45 °C is observed. This shows a limitation of this specific configuration designed for 15 °C. Under these conditions, a larger heat exchanger size, combined with a higher cooling air mass flow rate, can be implemented to increase cooling capacity at additional capital costs.

Ultimately, if this technology option was deployed in challenging environments with ambient air temperature much higher than 30 °C, an alternative to augmenting the size of the heat exchanger is to increase the flue gas temperature entering the

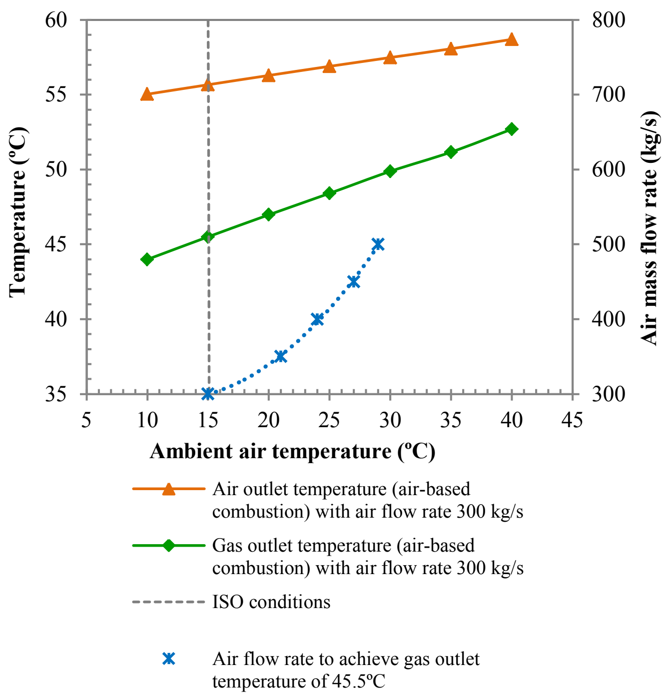

Fig. 9. Sensitivity to the ambient air temperature of the performance of a dry cooling configuration with two rotary heat exchangers in series for a conventional CCGT plant (without exhaust gas recirculation). Heating surface area 45868 m^2.

absorber at the expense of the energetic performance of the post-combustion capture unit.

It is worth noting that some wet cooling systems can also be affected by ambient conditions, notably if the evaporative cooling tower supplies water to the direct contact cooler.

7. Booster fan implications: power requirements and leakage control

An important consideration for this application of rotary heat exchangers is the management of gas leakage between flow streams. Leakage can be mitigated with strategies using proven purge and scavenge systems. Purge and scavenge flows can be extracted from the low-CO_2 reheated flue gas outlet ducting and, via a low leakage fan, passed back, under positive pressure, to specific purge and scavenge slots. The scavenge flow reduces the entrained leakage that is contained in the elements as they rotate, and the purge flow pressurizes the radial and axial seals reducing the amount of direct leakage. Operation at low rotational speed also reduces the entrained leakage [21].

The magnitude and the effects of leakage are highly dependent on the location of the flue gas booster fan required to overcome the additional pressure in the gas pathway – ducts, coolers and the packing of the absorber tower – and, also, avoid any negative effect of increasing the back pressure on the gas turbine exhaust.

In a standard wet cooled configuration, the booster fan can be located upstream of the direct contact cooler for a more accurate control of temperature at the absorber inlet. On the other hand, locating the booster fan downstream of the DCC offers the advantage of a lower temperature and volumetric flow rate at the inlet of the fan, which results in lower fan power consumption [9].

With rotary heat exchangers, leakage levels, the direction of the leakage flow, the power consumption and the temperature rise due to compression play an important role in the selection of the booster fan location. Locating the fan upstream of the regenerative heat exchanger, as shown in Figs. 2 and 3, is preferable for carbon capture since the temperature rise due to the compression occurs before the cooling process, resulting in a larger temperature pinch at the cold end of the rotary heat exchangers, reducing the element heating surface and the size of the heat exchanger. This effectively translates into lower pressure drop and lower booster fan power consumption.

Although the pressure differential between the gas streams results in leakage flow from high CO_2 concentration streams to low CO_2 concentration streams (i.e. the gas stream leaving the absorber or the stream of cooling air), and higher leakage levels compared to a configuration with the fan located directly upstream of the absorber tower, the overall magnitude of leakage is, however, very likely to be acceptable. The operating temperature differentials and resulting thermal deformations of the rotor would be significantly lower than power plant air preheaters or FGD gas heaters therefore better sealing can be expected. There is obviously a compromise between leakage and the capital costs of the device. A sensitivity analysis of leakage level on the overall CO_2 removal rate, illustrated in Fig. 10, shows that, at values lower than 1%, there is a marginal reduction of the overall CO_2 removal rate. The reduction is of the order of 0.45% for a leakage level of 0.5% for an absorber design with a 90% removal rate. It is also worth noting that leakage levels lower than 3% do not have any significant impact on the thermal performance analysis of the different configurations of rotary heat exchangers and results presented.

The total power consumption for each configuration is shown in Fig. 11 for a convectional CCGT plant and a CCGT plant with 40% EGR. Results include the booster fan, air fan and water pumps power and are reported for one single gas turbine – HRSG – absorber train of the two gas turbines in the reference plant. More details on the power consumption associated with the cooling water pumps of the wet and the hybrid system is provided in Appendix B, Table B.1.

The pressure drop through the gas pathway is of the same order of magnitude in all the configurations and, overall, gas/gas heat exchangers do not introduce additional pressure drops in the system. The trisector gas/gas/air heat exchanger configuration has the smallest overall dimensions and lowest booster fan power consumption.

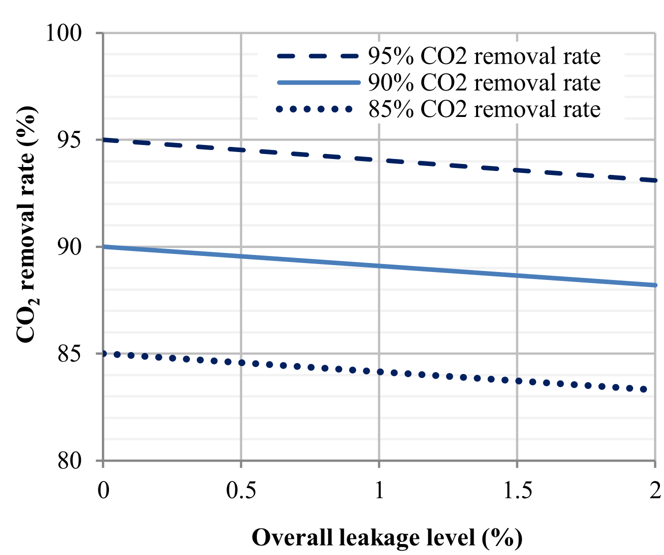

Fig. 10. Sensitivity of the overall CO_2 removal ratio to leakage level from the untreated flue gas into the CO_2-depleted gas.

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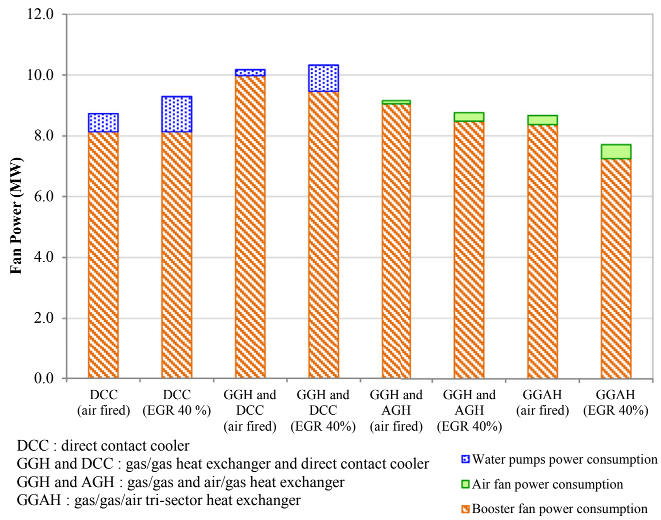

Fig. 11. Booster fan, air fan, cooling and process water pumps power consumption per gas turbine train for convectional (air-based combustion) combined cycle gas turbine plant and a plant with 40% exhaust gas recirculation ratio, both with post-combustion capture technology.

With exhaust gas recirculation, the reduction in gas flow rates and the pressure drops of the CO_2-depleted gas side lead to a reduction in booster fan power.

Finally, it is worth noting that the power consumption of air fans in the dry cooling systems is considerably smaller than the power of booster fans, and equivalent in magnitude to the power of cooling water pumps in wet and hybrid configurations.

8. Conclusions

The addition of post-combustion carbon dioxide capture systems increases cooling water demand in combined cycle gas turbine power plants. In a low carbon intensity electricity system with increased electricity use, the availability of cooling water and the impact on fresh water resources and marine environments need to be evaluated. The development of dry systems is likely to become widespread in water constrained environments.

This work investigates heat integration options transferring heat from the gas stream entering the absorber to the gas stream leaving the absorber, with the aim of reducing process and cooling water requirements of the direct contact cooler, whilst ensuring that the flue gas reaches appropriate temperature at the stack. Rotary regenerative gas/gas heat exchanger technology was considered for this purpose.

Three configurations using rotary heat exchangers are studied for a convectional CCGT plant and a CCGT with exhaust flue gas recirculation, and compared with the reference wet-cooling system. For the reference CCGT plant in this work, cooling water in the direct contact cooler is of the order of 30% of the total cooling water used in the rest of the plant, mostly in the condenser of the steam cycle.

Hybrid cooling systems, consisting of a gas/gas heat exchanger and a direct contact cooler, reduce cooling and process water usage by 67% and 35% respectively and ensure that an appropriate stack temperature in excess of 70 °C and an appropriate absorber inlet temperature around 45 °C can be achieved.

Dry air-cooled systems can be implemented to replace completely the direct contact cooler, eliminating the need for water usage to cool the flue gas prior to the absorber. Water savings with the removal of the direct contact cooler consist of water withdrawal from a fresh water source in a once-through cooling system, and water consumption in a recirculated and hybrid cooling system. A configuration with two rotary heat exchangers in series and a configuration with a single gas/gas/air heat exchanger with a trisector arrangement show good thermal performance for an air flow rate of 300 kg/s. A dry cooling system with a trisector configuration reduces complexity, the size and, potentially, the capital cost compared with a configuration with two rotary heat exchangers in series. For both configurations, pressure drop in the system is not significantly increased compared to a wet cooling configuration. The power consumption of the air fan is relatively small and equivalent in magnitude to the power of cooling water pumps in wet and hybrid configurations.

Dry cooling systems are highly dependent on ambient air temperature. For the air/gas heat exchanger designed for 15 °C, larger cooling air flow rates can compensate the increase on exhaust flue gas temperature at higher ambient temperatures up to 30 °C. Above this limiting value larger heat exchanger size, combined with a higher cooling air mass flow rate, need to be implemented.

In CCGT plants with exhaust gas recirculation of 40%, the higher absolute humidity of the exhaust flue gas leaving the heat recovery steam generator results in larger cooling and process water demand to cool the gas down to 45 °C at the absorber inlet, in both wet and hybrid cooling systems. In dry air-cooled systems, the flow rate of the CO_2-depleted gas leaving the absorber decreases, which, together with the larger amount of heat that needs to be removed from the gas stream, results in higher cooling air flow rates of 450 kg/s and 400 kg/s for a configuration with two rotary heat exchangers in series and with a gas/gas/air trisector heat exchanger, respectively.

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Appendix A. Reference natural gas combined cycle power plant

The reference plant is an 800 MWe air-fired NGCC (natural gas combined cycle) with a 2-on-1 configuration: two GE937IFB gas turbines with the flue gas exiting into two HRSGs, which jointly supply steam to a subcritical triple pressure steam cycle. The gas turbine operates at a TIT (turbine inlet temperature) of 1371 °C and AFR (air fuel ratio) of 40.5 on mass basis, at ISO atmospheric conditions and 100% load, with a power output at coupling of 285 MWe. In a combined cycle, the remaining heat contained in the exhaust flue gas is partially recovered to generate steam at three pressure levels. In this work, the flue gas is assumed to leave the HRSG at a temperature of 86 °C [9]. Two post-combustion capture units with primary amine-base solvent technology are implemented at the tail-end, one per each gas turbine - HRSG train. In each train, flue gas is cooled down before entering the absorber. The gas composition at the outlet is then calculated assuming a removal ratio of CO_2 of 90% and that gas leaves the water wash section saturated at a temperature of 45 °C. Stream variables are shown in Tables A.1 and A.2 for a single gas turbine – heat recovery steam generator – absorber unit train. The electricity output penalty of capture and compression in the air-fired CCGT and the CCGT with EGR is not considered in the scope of this work.

Table A.1
Exhaust flue gas stream information at HRSG outlet at different recirculation ratios (per GT-HSRG-absorber train).

EGR ratio		0	10	20	30	40
Temperature	°C	86	86	86	86	86
Pressure (HRSG outlet)	kPa	101.3	101.3	101.3	101.3	101.3
Dew Temperature (HRSG outlet)	°C	43.6	45.3	46.8	48.2	49.6
Pressure (fan outlet)	kPa	111.3	111.3	111.3	111.3	111.3
Dew Temperature (fan outlet)	°C	45.4	47.1	48.6	50.1	51.4
Mass flow rate	kg/s	670	662	655	648	643
Molar flow rate	mol/s	2353	2329	2305	2284	2263
Molar mass	g/mol	28.48	28.44	28.41	28.39	28.4
Composition						
CO_2	%vol	4.15%	4.62%	5.21%	5.97%	6.99%
H_2O	%vol	8.79%	9.57%	10.35%	11.12%	11.89%
N_2	%vol	74.26%	74.00%	73.82%	73.78%	73.93%
O_2	%vol	11.90%	10.92%	9.74%	8.25%	6.30%
Ar	%vol	0.89%	0.89%	0.88%	0.88%	0.89%

Table A.2
CO_2-depleted gas stream information at absorber outlet, at different recirculation ratios (per GT-HSRG-absorber train).

EGR ratio		0	10	20	30	40
Temperature	°C	45	45	45	45	45
Pressure	kPa	106.3	106.3	106.3	106.3	106.3
Mass flow rate	kg/s	633	556	482	410	340
Molar flow rate	mol/s	2264	1990	1726	1471	1225
Molar mass	g/mol	27.97	27.94	27.9	27.85	27.79
Composition						
CO_2	%vol	0.43%	0.48%	0.55%	0.65%	0.77%
H_2O	%vol	9.52%	9.56%	9.62%	9.70%	9.80%
N_2	%vol	76.82%	77.57%	78.53%	79.78%	81.51%
O_2	%vol	12.31%	11.45%	10.36%	8.92%	6.94%
Ar	%vol	0.92%	0.93%	0.94%	0.96%	0.98%

Appendix B. Assumptions

Table B.1
Assumptions.

Ambient air ISO conditions		
Temperature	°C	15
Pressure	bar	1.013
Relative humidity	%	60
Direct contact cooler		
Approach temperature between ambient wet bulb temperature and cooling water supply [9]	°C	7
Cooling water supply temperature	°C	15
DCC Bottom temperature differential with air cooling [9]	°C	46
DCC Bottom temperature differential with EGR	°C	23
DCC Cooler temperature pinch [9]	°C	7
Cooling water temperature rise [9]	°C	10
Pumps		
Efficiency [22]	%	75
Process water pump head [15,22]	m	57
Cooling water pump head [15,22]	m	26
Pump electric power to cooling duty ratio	%	0.8
Fans		
Efficiency	%	85
Pressure drop		
Absorber pressure drop [9,13]	kPa	5
Ducts pressure drop	kPa	2.5
Direct contact cooler pressure drop [9,13]	kPa	3
Rotary heat exchangers [23]		
Rotational speed	rpm	0.75–1
Element metal thickness	mm	0.75
Enamel total thickness	mm	0.3

Appendix C. Leakage flow rate and pressure drop fundamental equations

Leakage flow rate

A detailed calculation method is followed in the Howden's proprietary software to evaluate the total leakage level. Direct leakage flow rate is proportional to the heating surface area and the pressure differential across gasps. Entrained leakage flow rate is proportional to the void volume of the rotor and the rotational speed. Equation (C.1) shows a general expression for the direct leakage flow rate [20].

$$\dot{m}_{leakage} = K \cdot A \cdot \left(2g_c \cdot \Delta P_{gap} \cdot \rho_g\right)^{1/2} \tag{C.1}$$

Pressure drop

The pressure drop through the heater is a combination of duct losses at inlet of the rotor, expansions losses at outlet of the rotor and rotor losses. The pressure drop is proportional to the square of the fluid velocity and the density as illustrated in Equation (C.2) and (C.3), where K' is the proportional constant [20]. Detailed calculation method is followed in Howden's proprietary software.

$$\Delta P = K' \cdot \frac{V_g^2}{2} \cdot \rho_g \tag{C.2}$$

$$\Delta P = K' \cdot \frac{\left(\dot{m}_g / A_n\right)^2}{2 \cdot \rho_g} \tag{C.3}$$

Nomenclature

A	Flow area (m^2)
A_n	Cross-sectional area open to flow (m^2)
g_c	Gravitational constant (1 kg m/N s^2)
K	Discharge coefficient, dimensionless (generally 0.4 to 1.0)
K'	Proportionality constant (s^2/m)
$\dot{m}_{leakage}$	Leakage mass flow rate (kg/s)
$\dot{m}$	Mass flow rate of the fluid (kg/s)
ΔP	Pressure drop (kg/m^2)
ΔP_{gap}	Pressure differential across gap (kg/m^2)
V	Velocity of the fluid (m/s)
ρ	Density of the fluid (kg/m^3)

Subscript

g	Gas

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